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To cite this article: David Valentin *et al* 2017 *J. Phys.: Conf. Ser.* **813** 012036

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Dynamic response of the MICA runner. Experiment and simulation

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Abstract. Studying the dynamic response of Francis turbine runners is of paramount importance in order to avoid resonance problems during operating conditions. For this purpose, the natural frequencies as well as their associated mode-shapes and damping ratios of the runner have to be determined. In this paper, an Experimental Modal Analysis (EMA) of the runner of a Francis turbine prototype has been performed. By means of this experimental technique, natural frequencies, mode-shapes and damping ratios have been estimated in air. Results obtained have been compared with a Finite Element Method (FEM) model in order to check the accuracy of the simulation.

1. Introduction

The structural dynamic response of Francis turbines has to be studied during the stage of design of the machine in order to avoid resonance and fatigue problems during operating conditions [1, 2]. One of the components that could be strongly affected by fatigue problems is the runner. The runner is a rotating structure submerged and confined in water with small gaps to the stationary parts. The effect of the water and the confinement on the natural frequencies of the runner is well-known and it is demonstrated with the added mass theory [3-5]. This added mass effect could be accurately estimated by using acoustical-numerical Fluid Structure Interaction (FSI) simulations [5-8]. However, the damping factor associated to each natural frequency and mode-shape plays also an important role on the dynamic response of structures, since it is the mechanism of the structure to dissipate energy due to inertia and elastic forces.

The damping ratio associated to each natural frequency and mode-shape can be understood as a sum of the material damping ratio and the hydrodynamic damping ratio or added damping due to the surrounding water [3, 9, 10]. This hydrodynamic damping has been demonstrated to be higher when the submerged structure is near a rigid wall [3, 10] and even higher when the surrounding water flows through the submerged structure [9]. The estimation of this damping ratio using simulation is difficult and challenging and therefore experimental results are usually necessary.

The mode-shapes associated to each natural frequency of the structure has to be also studied in detail to understand the dynamic response of the machine under operating conditions. In case of resonance during operation, not only the frequency of the excitation has to coincide with a natural frequency of the runner, but also the shape of the excitation has to match with the mode-shape of the natural frequency [11]. According to many studies, confining the structure in still water does not change its mode-shapes [3, 5]. However, when the runner rotates in water every mode-shape became



into two travelling waves rotating in the same and in the opposite direction than the runner but maintaining the same shape than without rotation [12].

In this paper, an Experimental Modal Analysis (EMA) of a medium-head Francis turbine has been carried out. The Francis turbine selected for the study is located in British Columbia, Canada, and it has 444 MW of rated power. The study of the dynamic response of this prototype is included in the research carried out in the Hyperbole project. Natural frequencies, damping ratios and mode-shapes of the runner in air have been determined and compared with simulation results.

2. Experiment

Taking profit of an outage of the prototype in 2016, an EMA could be performed in the runner. The runner was accessible from the draft tube mandoor as well as from the spiral casing. Impacts were carried out with an instrumented hammer (Dytran 5802A, sensitivity 220 uV/N) in different points of the runner and the response was measured with accelerometers located always in the same point. This method is called rowing hammer method. The runner was impacted at 16 different equidistant points in the band outlet and in every blade outlet. One accelerometer (Kistler 8750A50, sensitivity 100 mV/g) was located in the band outlet and another two in the crown and band inlet (see Figure 1).

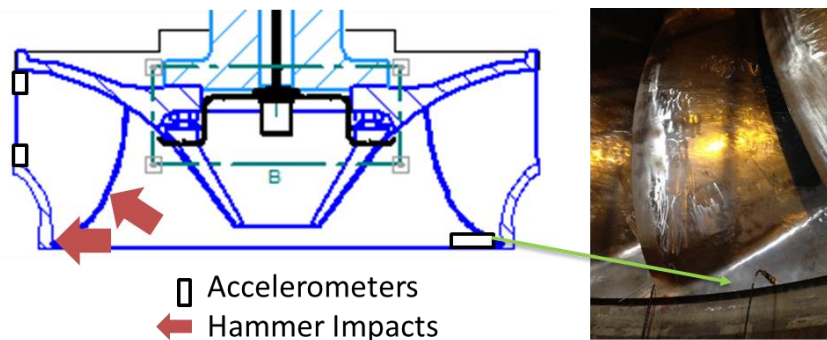


Figure 1. Instrumentation and impacts location.

Five impacts were always done in every point to use their average during the analysis. An exponential window ($\tau=400$ ms) was applied to the accelerometers time signal and a transient window (100ms) to the hammer signal for the FRF (Frequency Response Function) computation. Coherence between the accelerometers signals and the hammer was computed always to ensure that the response is due to the impact. Figure 2 shows an example of one FRF computed at the same time than the coherence. It is seen that for all the peaks found in the FRF, which are the natural frequencies of the runner, the coherence is maximum.

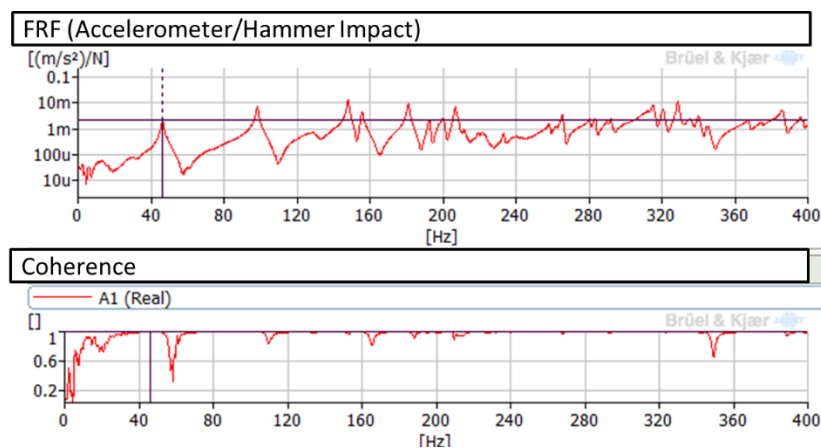


Figure 2. Example of a FRF and its coherence.

This procedure was repeated for all the different points (32) where the runner was impacted. To know how the mode-shapes of the structure are, all these points were assigned to a simplified structure which represents the runner performing ODS (Operation Deflection Shape). Natural frequencies and damping ratios were extracted for every mode using Complex Mode Indicators Functions (CMIF) of the commercial software Pulse® Reflex [13].

3. Numerical model

The numerical model considered the runner, the shaft and the generator. However, as only impacts in the runner were done experimentally, only the runner mode-shapes are shown in this paper. The details of the optimal mesh and the different parameters fixed in the FEM simulation were previously published in [14]. The FEM simulation was solved using the commercial code Ansys® v16.2. In this case, the surrounding water of the turbine is not considered in order to compare the simulation with the experimental case. Moreover, this simulation does not consider any damping in the numerical model, therefore this parameter will not be compared with experimental results.

4. Results

As a first view, comparing the response of different points of the runner (band inlet, band outlet and blades outlet, Figure 3), it is clearly seen that the response of the band and the blades at the outlet is rather higher than the response in the inlet. This means that maximum displacement of the mode-shapes is basically concentrated in the band and blades outlet.

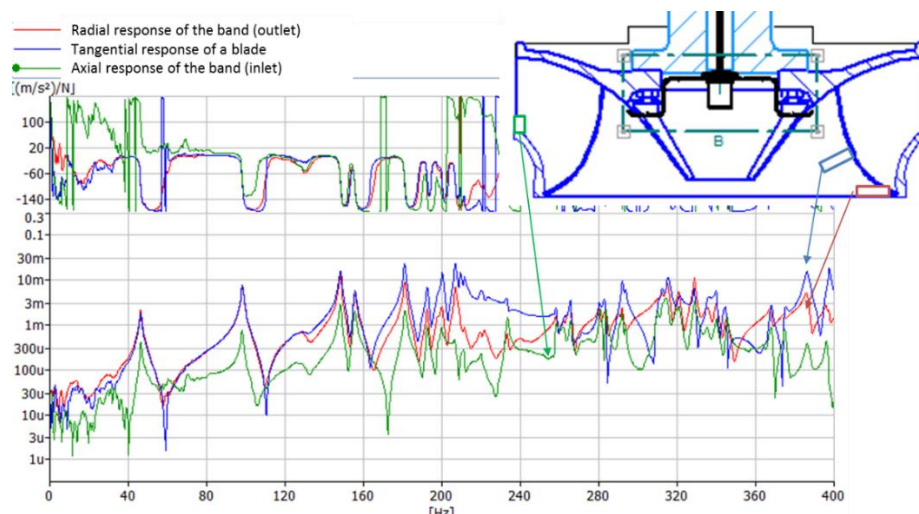


Figure 3. Runner natural frequencies within the frequency range 0-400 Hz. Response of the structure in different points (Green=response in the band inlet, Blue=response in the blade outlet, red=response in the band outlet).

The mode-shapes of Francis turbine runners are normally classified by the number of nodal diameters (iND) [14], but these different mode-shapes with different number of nodal diameters have also another common characteristics that can be used to recognise them. Figure 4 shows which are these common characteristics between the mode-shapes of the Francis turbine of study. There is a first zone in frequency (from 0 to 160 Hz) where the amplitude of the band and blades outlet is the same, these modes have been named as Global Modes (G). However, from 160 Hz onwards, the blade deformation is higher than the band deformation and therefore these modes have been named as Blade Dominant (BID) modes. Moreover, analysing the deformation and the phases of two consecutive blades (Blades number 15 and 16), there are some modes where both are in phase (IPh), and another in counter phase (CPh) (as seen in Figure 4). Therefore, different combinations of nodal diameters (iND), Global Modes or BID, or IPh or CPh modes can be found for this Francis turbine runner.

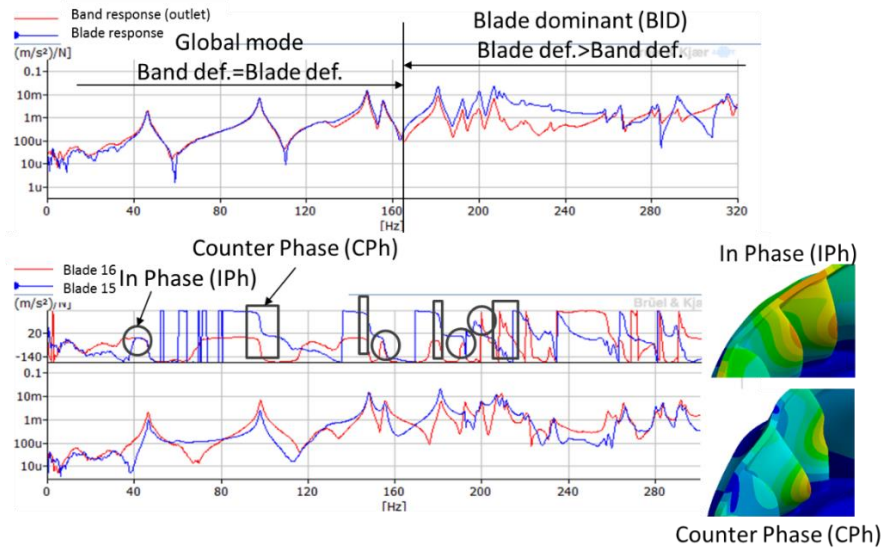


Figure 4. Classification of the mode-shapes.

Table 1. First mode-shapes found within the frequency band 0-210 Hz.

Mode-Shape	f_n (Hz) Experiment	f_n (Hz) Simulation	Difference Exp-Sim (%)	Damping ratio (%)
2ND-G-IPh	46.44	44.27	4.66	1.0272
3ND-G-IPh	98.24	93.6	4.73	0.5498
1ND-G-IPh	129.14	126.28	2.22	2.4669
4ND-G-CPh	148.08	140.65	5.02	0.3094
2ND-BID-IPh	155.58	152.44	2.02	0.3469
5ND-BID-CPh	181.03	171.34	5.35	0.2713
3ND-BID-IPh	192.48	189.06	1.77	0.2394
2ND-BID-CPh	199.70	188.54	5.59	0.2894
0ND-BID-IPh	206.85	202.49	2.11	0.2490
1ND-BID-CPh	209.38	196.86	5.98	0.2413

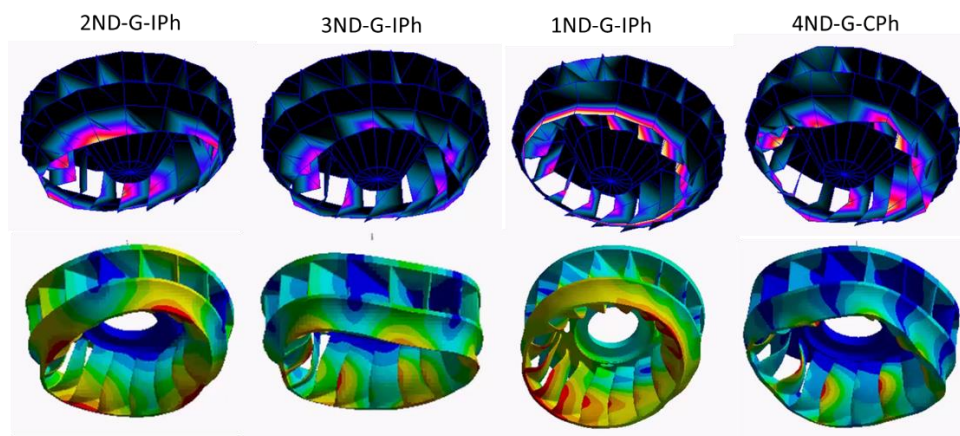


Figure 5. ODS of the first global mode-shapes.

Table 1 shows the value of the natural frequencies and damping ratio for the first ten mode-shapes found for the runner of study. Good agreement between experimentation and simulation is found for the natural frequencies values as well as for the mode-shapes. A comparison between the ODS obtained by experimentation and simulation is also shown in Figure 5.

5. Conclusions

Natural frequencies, damping ratios and mode-shapes of a Francis turbine runner have been obtained by means of an Experimental Modal Analysis. The runner was impacted in 32 different points and the response was measured with accelerometers in different positions. After the FRF computation an ODS analysis was performed and the mode-shapes, natural frequencies and damping ratios were obtained. These results have been compared with simulation having a good agreement.

This experimental analysis validates the numerical simulation and therefore the numerical model can be used to obtain the natural frequencies when the runner is surrounded by water.

References

- [1] Egusquiza, E., Valero, C., Huang, X., Jou, E., Guardo, A., and Rodriguez, C., 2012, "Failure investigation of a large pump-turbine runner," *Engineering Failure Analysis*, 23(0), pp. 27-34.
- [2] Ohashi, H., 1994, "Case Study of Pump Failure Due to Rotor-Stator Interaction," *International Journal of Rotating Machinery*, 1(1), pp. 53-60.
- [3] Valentín, D., Presas, A., Egusquiza, E., and Valero, C., 2014, "Experimental study on the added mass and damping of a disk submerged in a partially fluid-filled tank with small radial confinement," *Journal of Fluids and Structures*, 50(0), pp. 1-17.
- [4] Askari, E., Jeong, K.-H., and Amabili, M., 2013, "Hydroelastic vibration of circular plates immersed in a liquid-filled container with free surface," *Journal of Sound and Vibration*, 332(12), pp. 3064-3085.
- [5] Liang, Q. W., Rodríguez, C. G., Egusquiza, E., Escaler, X., Farhat, M., and Avellan, F., 2007, "Numerical simulation of fluid added mass effect on a francis turbine runner," *Computers & Fluids*, 36(6), pp. 1106-1118.
- [6] Rodriguez, C. G., Flores, P., Pierart, F. G., Contzen, L. R., and Egusquiza, E., 2012, "Capability of structural-acoustical FSI numerical model to predict natural frequencies of submerged structures with nearby rigid surfaces," *Computers & Fluids*, 64(0), pp. 117-126.
- [7] Jacquet-Richardet, G., and Dal-Ferro, C., 1996, "Reduction method for finite element dynamic analysis of submerged turbomachinery wheels," *Computers & Structures*, 61(6), pp. 1025-1036.
- [8] Valentín, D., Presas, A., Egusquiza, E., and Valero, C., 2016, "On the Capability of Structural-Acoustical Fluid-Structure Interaction Simulations to Predict Natural Frequencies of Rotating Disklike Structures Submerged in a Heavy Fluid," *Journal of Vibration and Acoustics*, 138(3), pp. 034502-034502.
- [9] Monette, C., Nennemann, B., Seeley, C., Coutu, A., and Marmont, H., 2014, "Hydro-dynamic damping theory in flowing water," *IOP Conference Series: Earth and Environmental Science*, 22(3), p. 032044.
- [10] Naik, T., Longmire, E. K., and Mantell, S. C., 2003, "Dynamic response of a cantilever in liquid near a solid wall," *Sensors and Actuators A: Physical*, 102(3), pp. 240-254.
- [11] Presas, A., Egusquiza, E., Valero, C., Valentin, D., and Seidel, U., 2014, "Feasibility of Using PZT Actuators to Study the Dynamic Behavior of a Rotating Disk due to Rotor-Stator Interaction," *Sensors*, 14(7), pp. 11919-11942.
- [12] Presas, A., Valentin, D., Egusquiza, E., Valero, C., and Seidel, U., 2015, "Influence of the rotation on the natural frequencies of a submerged-confined disk in water," *Journal of Sound and Vibration*, 337(0), pp. 161-180.
- [13] Binda, M., Harčarik, T., and Šimčák, F., "Modal parameters estimation by using PULSE Reflex modal analysis," *Proc. 4th International Conference of Modelling of Mechanical and Mechatronic Systems*, pp. 17-21.
- [14] Valentín, D., Ramos, D., Bossio, M., Presas, A., Egusquiza, E., and Valero, C., "Influence of the boundary conditions on the natural frequencies of a Francis turbine," *Proc. IOP Conference Series: Earth and Environmental Science*, IOP Publishing, p. 072004.